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FOREWORD

This program was funded by the U. S. Army Air Mobility Research and Development Laboratory. Program management was by the Lewis Research Center of the National Aeronautics and Space Administration under Contract NAS 3-20045. The period of performance was 1 July 1976 to 30 April 1977.

Technical direction was provided by the NASA Project Manager, Mr. Lawrence P. Ludwig of the Fluid Systems Components Division. Mr. Leonard W. Schopen, NASA Lewis Research Center, was the Contracting Officer.

The Avco Lycoming Program Manager was Mr. Peter Lynwander. Mr. Michael O'Brien was the principal investigator.

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SUMMARY

An experimental evaluation was performed on a synergistic type shaft seal incorporating features intended to improve its performance. The need for these improvements was determined during a previous program (Reference 1), which evaluated a similar seal. The seal is intended for use in advanced aircraft and helicopter transmissions and was tested at conditions simulating those applications.

The seal design incorporates features of both elastomeric lip seals and segmented carbon seals. A moulded elastomeric retainer similar to a lip seal elastomeric element is used to hold and position the segmented carbon sealing rings. The retainer and carbon ring assembly is held in place by means of a pinch plate.

Testing was performed at bearing cavity pressures up to $.34 \text{ N/cm}^2$ (.5 psi) and surface velocities to 72.9 m/s (14,349 ft/min). Static shaft runout was maintained at $.0038 \text{ cm}$ (.0015 in.). Seal oil leakage, shaft runout, and torque were measured during all dynamic testing.

The seal initially exhibited a slight but not excessive rate of leakage (.47 cc/hr) at 0.0 N/cm^2 bearing cavity pressure and surface speed of 72.9 m/s (14,349 ft/min). Seal torque was observed at 2.4 N-m (21 in.-lb) at these conditions, which is considered to be an acceptable level. A series of modifications were made to improve the leakage characteristics at $.34 \text{ N/cm}^2$ (.5 psi) bearing cavity pressure, but these were not successful. The leakage is believed to be due to the lack of shaft contact exhibited by the oil side carbon element, swelling of the elastomeric retainer, and runner tracking problems. The observed oil leakage was found to be both speed and pressure dependent.

The anti-rotation lug problem encountered in the previous program (Reference 1) which tended to release the sealing elements during operation was overcome by redesign features incorporated in this test seal. Although the test seal appears to be structurally sound, additional development will be required to obtain a reliable sealing configuration particularly at positive bearing cavity pressures.

INTRODUCTION

Shaft seals are becoming increasingly critical components in advanced aircraft transmissions. The function of these seals is twofold; (a) to prevent lubricant escape and (b) to prevent ingestion of water and debris which may damage the gears and bearings. Seal problems are usually recognized when an excessive amount of oil leaks from the seal. This is usually not a flight safety problem, but excessive seal leakage increases both maintenance costs and aircraft down time. High surface speeds combined with internal gearbox pressure and shaft runout cause high heat generation, seal wear, and oil leakage. In order to avoid costly premature gearbox removals due to seal failure, the minimum life of these seals should be at least equal to the scheduled overhaul time of the transmissions.

Present helicopter transmission seals operate at speeds to 76.2 m/s (15,000 ft/min) with oil-to-air pressure differentials normally less than 0.689 N/cm^2 (1 psi) (Reference 2). For helicopter transmissions the sealed fluid is generally a synthetic oil (MIL-L-23699 or MIL-L-7808 at a maximum temperature of 394 K (250°F)). Aircraft accessory transmissions operate at somewhat higher temperature 450 K (350°F) and pressure 5.516 N/cm^2 (8 psi).

Modern transmissions use three basic types of shaft seals, an elastomeric lip seal, a labyrinth seal, or a mechanical seal with a carbon ring in sliding contact with the shaft or shaft shoulder.

Elastomeric lip seals are inherently limited in sliding speed capability for two reasons. First, as sliding speed increases, the temperature of the lubricant under the sealing lip increases, and modest sliding speeds can cause lubricant temperatures to reach the boiling point. These high local lubricant temperatures cause gradual degradation of the elastomer and subsequent seal failure. Second, as sliding speed increases, there is a decreasing capability of the elastomeric seal to follow the runout motion of the shaft, this results in leakage.

Labyrinth seals are sometimes used when sliding speeds exceed the capability of elastomeric lip seals. Labyrinth seals are not positive contact seals and, therefore, are prone to lubricant leakage, and debris and dirt are not excluded in a positive manner from the bearing and gear compartment.

Mechanical seals with carbon ring sealing elements have high-speed capability but their cost, as compared to elastomeric and labyrinth seals is commonly higher by an order of magnitude (or more). Then too, the mechanical seals of the face type have assembly requirements more complex than lip or labyrinth seals and require more space. Further, mechanical seals of the face type are prone to failure because of compatibility problems associated with the elastomeric secondary seal. The mechanical seals of the shaft riding type require very pre-

cise manufacturing tolerances, have very little ability to accommodate shaft angular misalignment, and have friction forces which inhibit the carbon rings from following the shaft runout motion; thus they are prone to leakage.

The purpose of this program was to continue the development of a synergistic seal similar to that tested in Reference 1.

APPARATUS AND PROCEDURE

Test Rig

The test vehicle was the T55-L-7 engine No. 2 position seal test rig (Figure 1), modified to accept the test seal. The test rig prime mover was an 8-inch, 74.57 kw (100 hp), 30,000 rpm steam turbine. An automatic speed control was used to maintain a constant test rig speed. The installation is shown in Figure 2. Lubrication of the test rig and test seal was provided by two separate oil systems as shown in the lubrication schematic (Figure 3). System I serviced the test seal and adjacent bearing while System II serviced the support bearing package.

The test rig was designed to allow varying degrees of seal runner runout to be built into the test rig. The seal runner was not piloted but instead was fastened to an adapter (clamped on the shaft) with bolts. The radial clearance remaining in the bolt holes allowed the runner to be shifted, and the desired runout was achieved. A proximity probe was used to monitor the dynamic shaft runout.

Oil feed temperature for bearing lubrication and seal cooling was maintained at 338 ± 5 K ($150 \pm 10^\circ\text{F}$). This temperature was selected to encourage leakage modes dependent upon higher lubricant viscosity. One direct impingement jet and one indirect impingement jet were used (Figure 4). Bearing lubricating oil flow was maintained at 136 kg/hr (300 lb/hr) throughout the test. Seal cooling flow was initially maintained at 84 kg/hr (185 lb/hr), but subsequently was redirected and increased to 136 kg/hr (300 lb/hr).

Recorded Parameters

The following pertinent parameters were observed and recorded at each test point.

- Seal Oil Leakage
- Seal Torque
- Dynamic Shaft Runout

Seal oil leakage was measure by collection in a graduated cylinder. Seal torque was indicated with a rotary torque tranducer.

The original test plan specified a 65-hour test of the seal at constant conditions, as described below:

Shaft Speed - 10,000 rpm 72.9 m/s (14,349 ft/ min)

Bearing Cavity pressure - 1.38 N/cm² (2 psi)

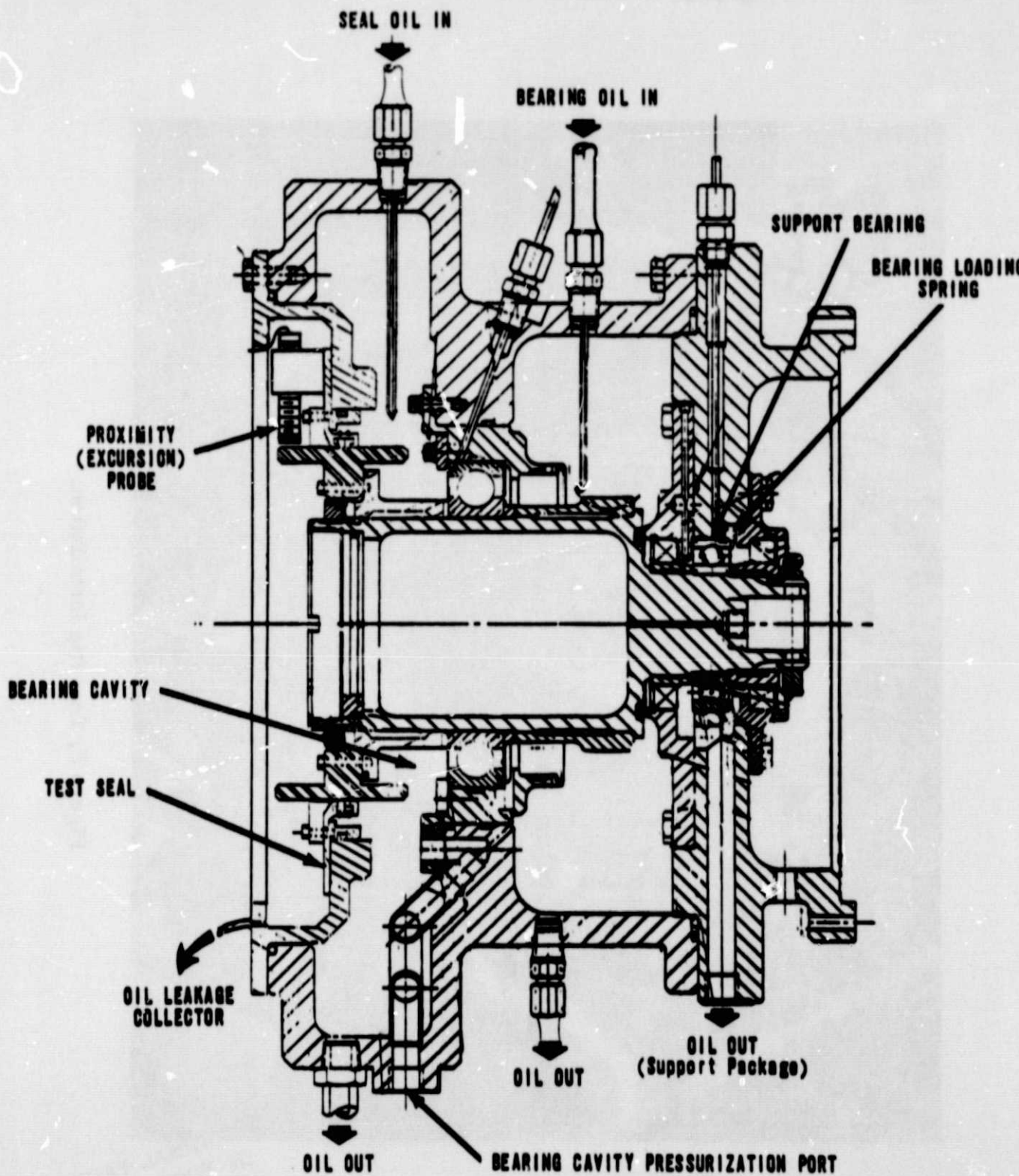


Figure 1. Test Rig.

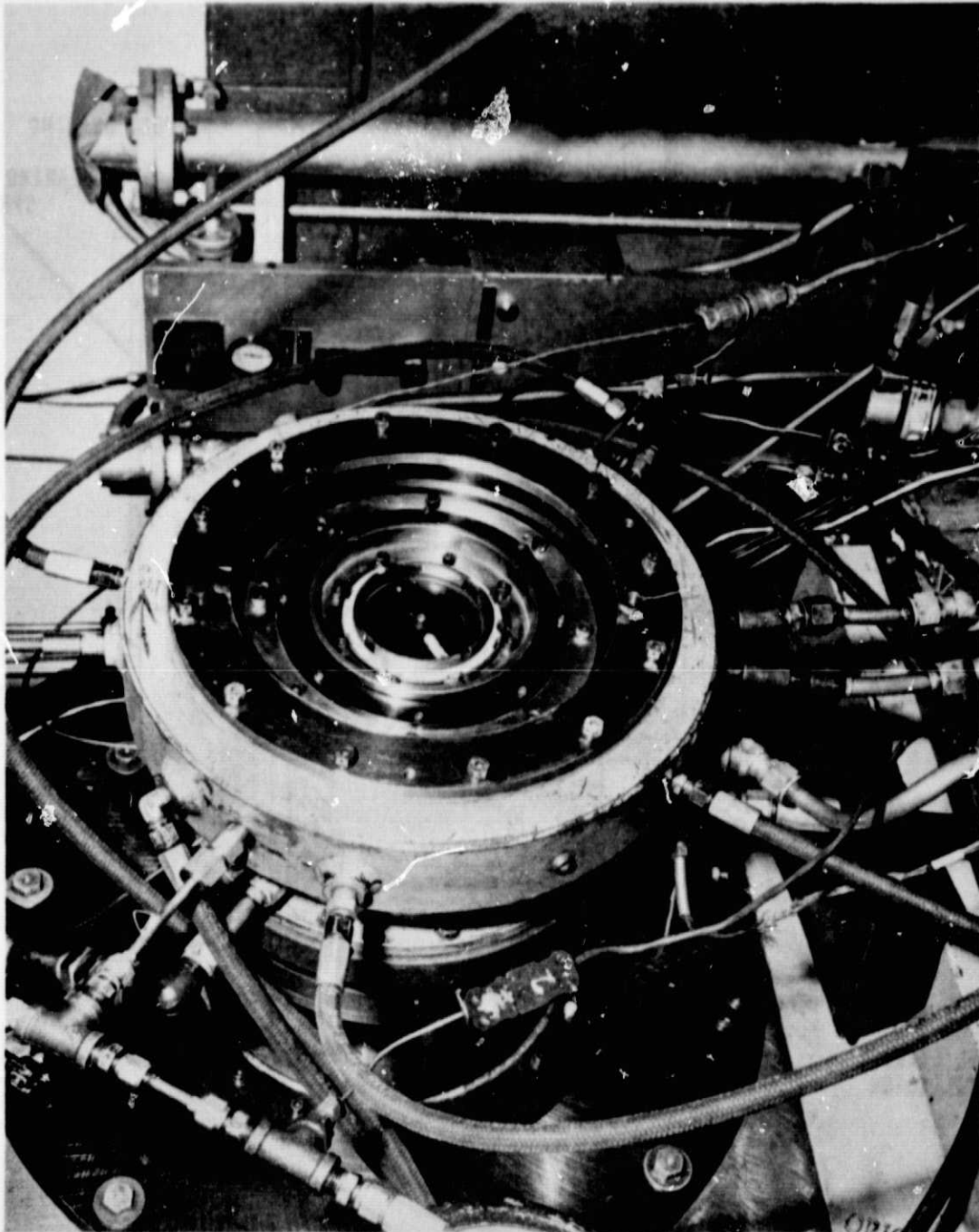


Figure 2. Test Rig Installation.

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SYSTEM II

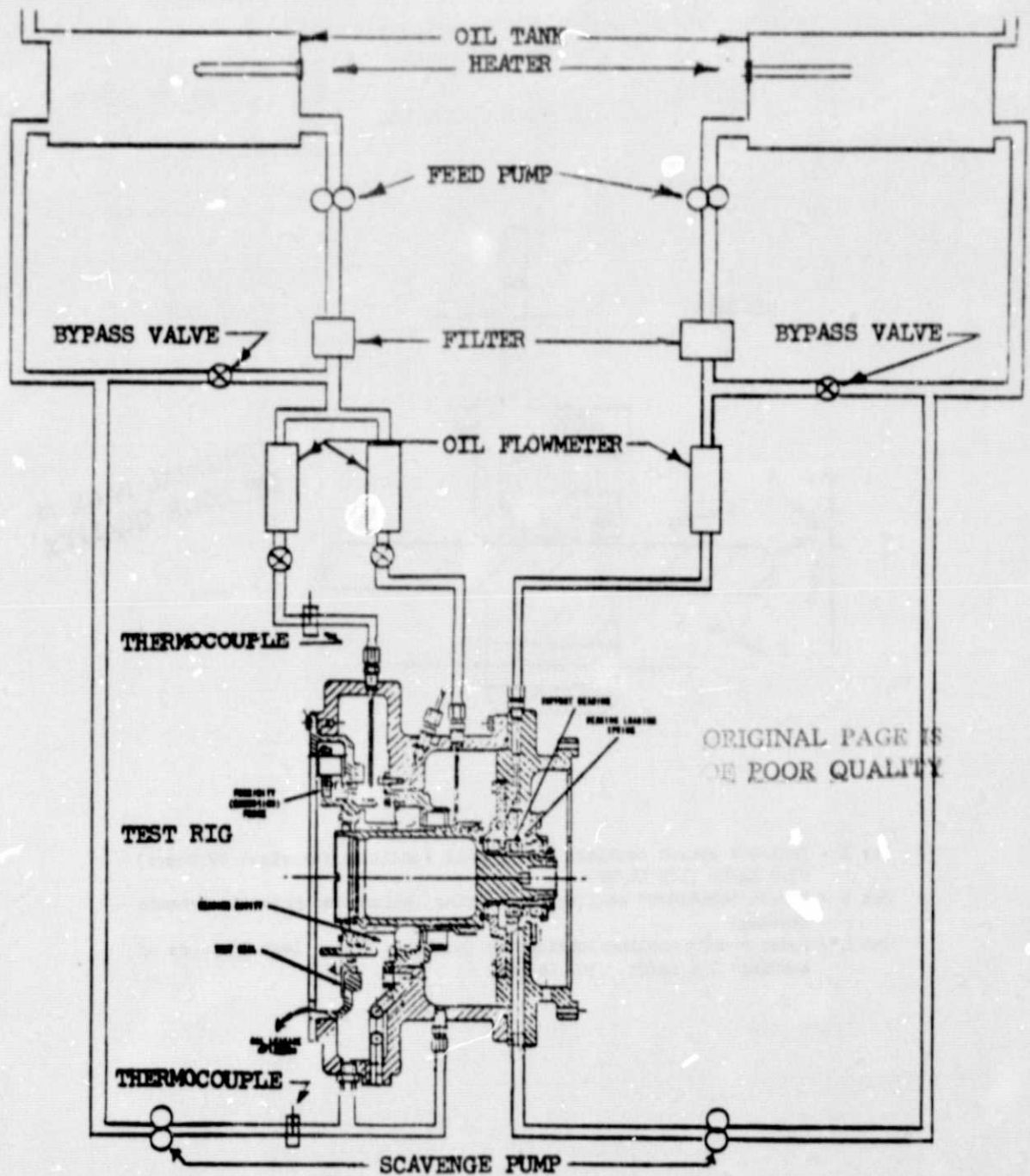
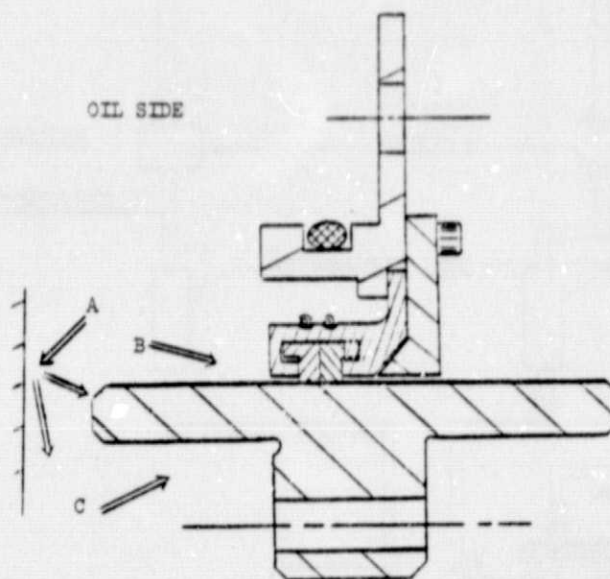


Figure 3. Schematic of the Lubrication System.



- Jet A - Indirect splash cooling/ lubricating (utilized for first 47 hours)
83.9 Kg/Hr (185 Lb/Hr)
- Jet B - Direct impingement cooling/lubricating (eliminated prior to Dynamic Testing)
- Jet C - Under runner cooling/lubricating (utilized during last 18 hours of testing) 136 Kg/Hr. (300 Lb/Hr)

Figure 4. Location of Seal Oil Jets.

Shaft Runout - .0102 cm (.004 in.)

Variations in the above test plan were made to investigate unusual seal performance.

Test Seal

Synergistic Seal Design

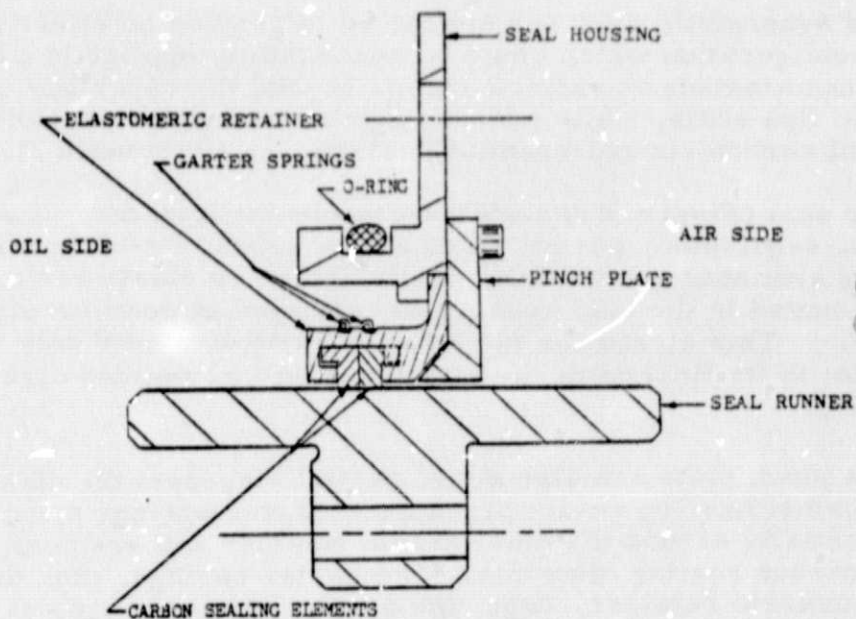
The synergistic seal was conceived to provide an effective, low-cost seal configuration which could be successfully applied in advanced aircraft transmissions at surface speeds beyond the capability of elastomeric lips seals, while addressing shortcomings encountered with conventional carbon circumferential seal designs (Reference 2).

The seal (Figure 5) includes six major components. Two interchangeable, segmented, carbon rings act as primary sealing elements. The sealing elements are held and positioned by an elastomeric retainer, which is mounted in the seal housing and clamped in position with a pinch plate. This elastomer mounting scheme was used only for testing convenience; in its final form the elastomer will be molded directly to a steel housing.

The pinch plate can also act to partially support the elastomeric retainer against bearing cavity pressure. Garter springs are mounted circumferentially around the elastomeric retainer and are positioned above the carbon sealing elements. The garter springs, plus the tension of the elastomeric retainer, urge the sealing elements against the seal runner. Sealing is accomplished by maintaining contact between the inside diameter of the sealing element and the runner and between the elastomeric retainer and the outside diameter of the sealing element. Staggering of the segmented rings provides blockage of the potential leakage paths at the segment end gaps. Anti-rotation provision for the carbon rings is made with two lugs machined on the carbon segments, which are accommodated in matching slots molded into the elastomeric retainer.

The following features, envisioned during the previous test program (Reference 1), were incorporated into the test seal to improve its performance.

1. Redesigned anti-rotation lugs were required to preclude the release of and subsequent spinning of the carbon rings during seal operation.
2. Higher durometer elastomer was used (increased from 60 to 75) to reduce deformation of the elastomer, particularly in the anti-rotation lug area.



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Seal Material

Elastomer - Viton
Housing - 316 Stainless Steel
Sealing Elements - Carbon Graphite
Garter Springs - 300 Series Stainless Steel
Seal Runner - AMS 6382 Steel
Seal Runner Surface - Chrome Plate - .00254 To .00762 CM
(.001 To .003 IN.) Thick
Per AMS 2106

Seal Design Data

Seal Runner O.D. $13.932 \pm .00127$ CM ($5.481 \pm .0005$ IN.)
Seal Runner Finish - 6 AA
Seal Runner Roundness - .00025 CM (.00010 IN.)
Garter Spring Load - 7.784 N (1.75 Lb) Initially, increased to
11.1 N (2.5 Lb)

Figure 5. Test Seal

3. A redesigned elastomeric retainer incorporated features to axially pinch the carbon sealing elements after assembly. This was to improve anti-rotation lug engagement and positioning of the carbon elements.

TEST PROGRAM

Initial Testing

Prior to the initiation of testing, photographs of test hardware were taken. Figures 6 through 8 show the following; Test seal components prior to test; assembled seal and runner prior to test; a close-up of the seal ID prior to test. A silicone grease, used to aid in assembly of the seal, is visible on the seal bore in Figure 8.

Test History

Prior to the start up of the test rig, static oil leakage (>2 cc/hr) was observed coincident with activation of the test rig oil system. Leakage occurred without imposing a bearing cavity pressure. In an attempt to reduce the leakage, garter spring tension was increased from 7.78 N (1.75 lb) to 8.89 N (2.01 lb). The seal runner runout was also adjusted to .0038 cm (.0015 in.). These modifications were unsuccessful in stopping the leakage. Since the leakage appeared to be in the vicinity of one of the seal cooling oil jets (one directed at the sealing interface), the oil flow to this jet was reduced, then eliminated. The oil jet impinging directly on the sealing interface apparently built up enough dynamic head to cause leakage. The indirect cooling oil jet was retained.

The seal was dynamically tested at 10,000 rpm 72.9 m/s (14,349 ft/min) with no bearing cavity pressure. Test duration was 8.5 hours, and the leakage rate was .47 cc/hr. Bearing cavity pressure was increased to .34 N/cm² (.5 psi) for 1.5 hours, yielding a leakage rate of 5.2 cc/hr.

Speed was reduced to 6600 rpm, 48.1 m/s (9,470 ft/min) in an attempt to reduce the 5.2 cc/hr leakage rate, with the following results:

Conditions: 6600 rpm, 48.1 m/s (9,470 ft/min);
.34 N/cm² (.5 psi);

First 3 hours - 3.2 cc/hr
Second 3 hours - 3.8 cc/hr
Third 3 hours - 2.1 cc/hr
Fourth 3 hours - 9.3 cc/hr

Overall leakage rate - 4.6 cc/hr

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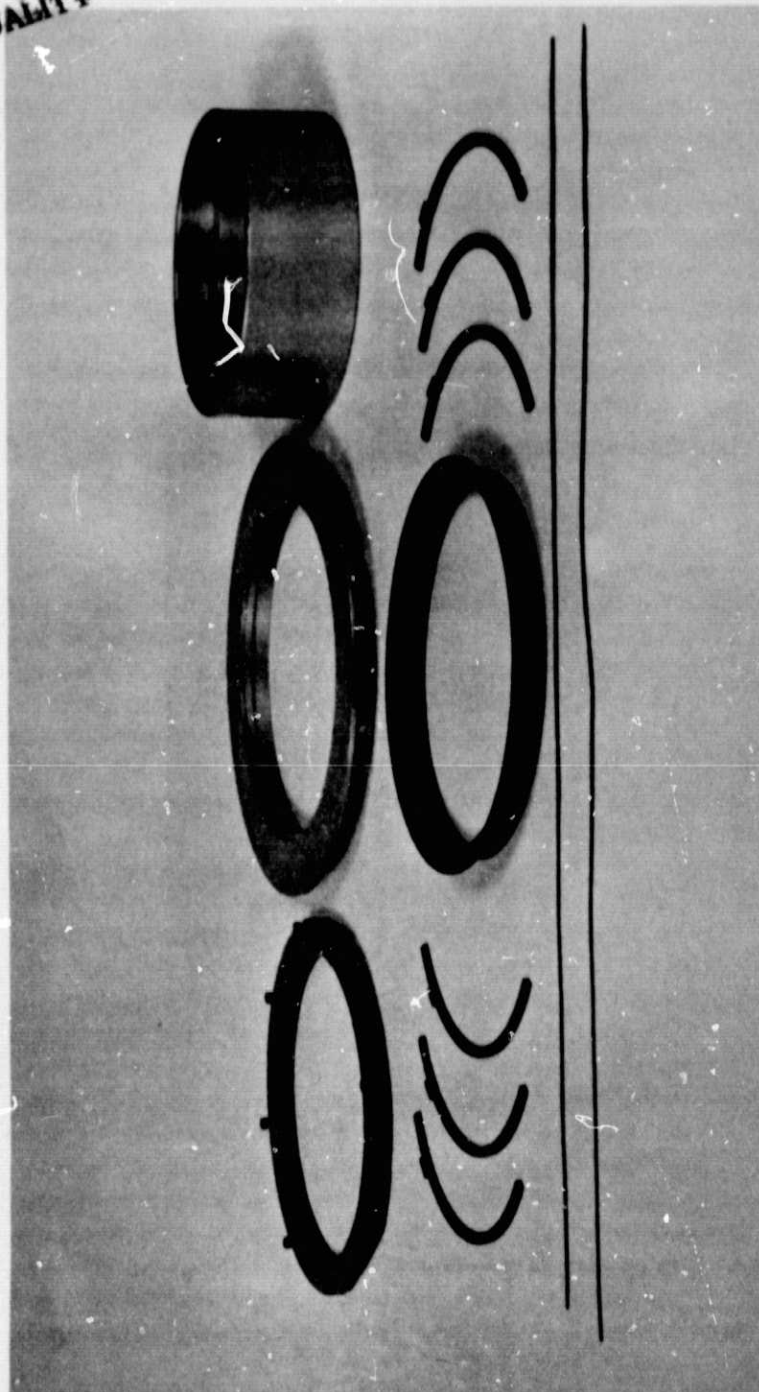


Figure 6. Test Seal Components Before Test.

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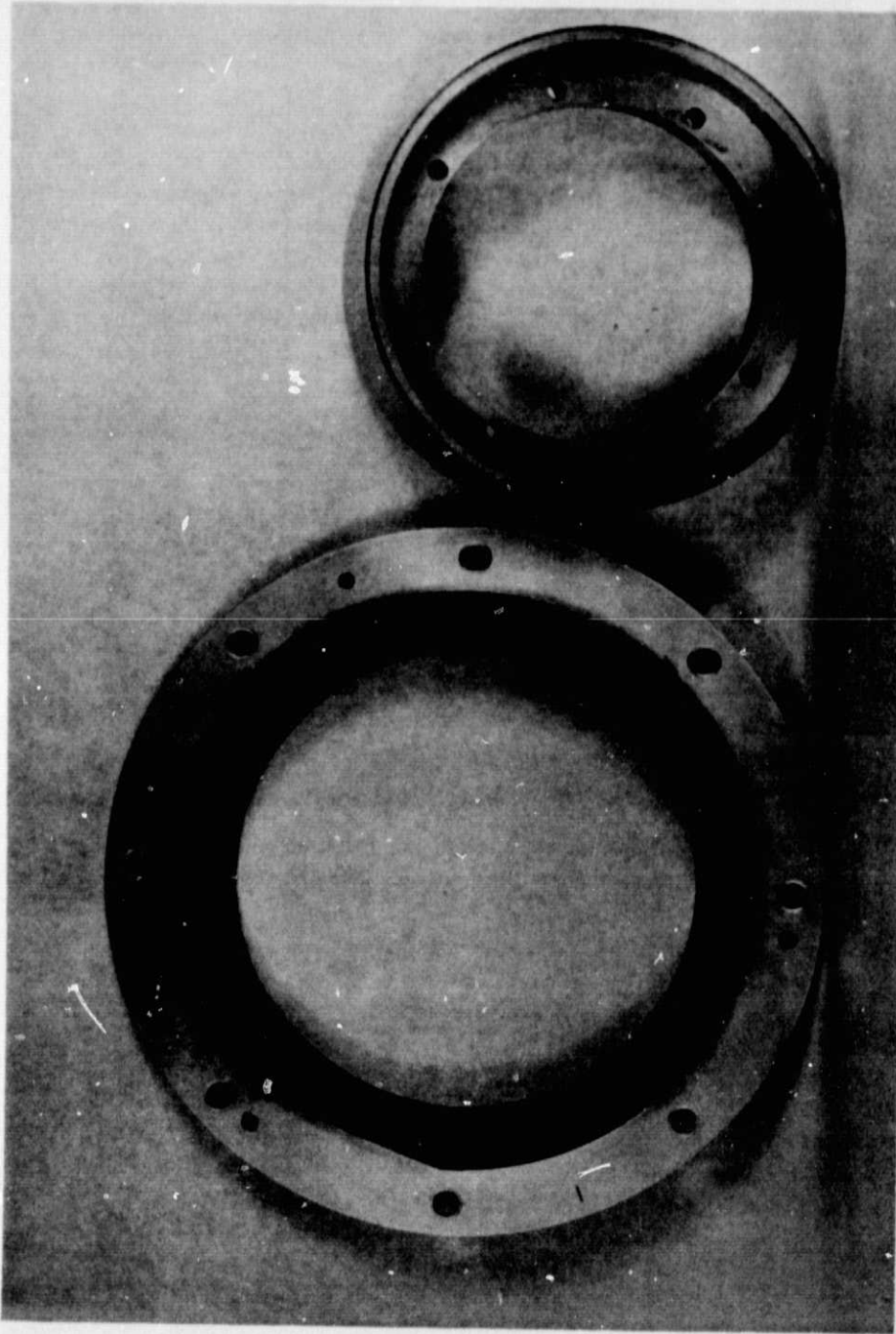


Figure 7. Assembled Seal Before Test.

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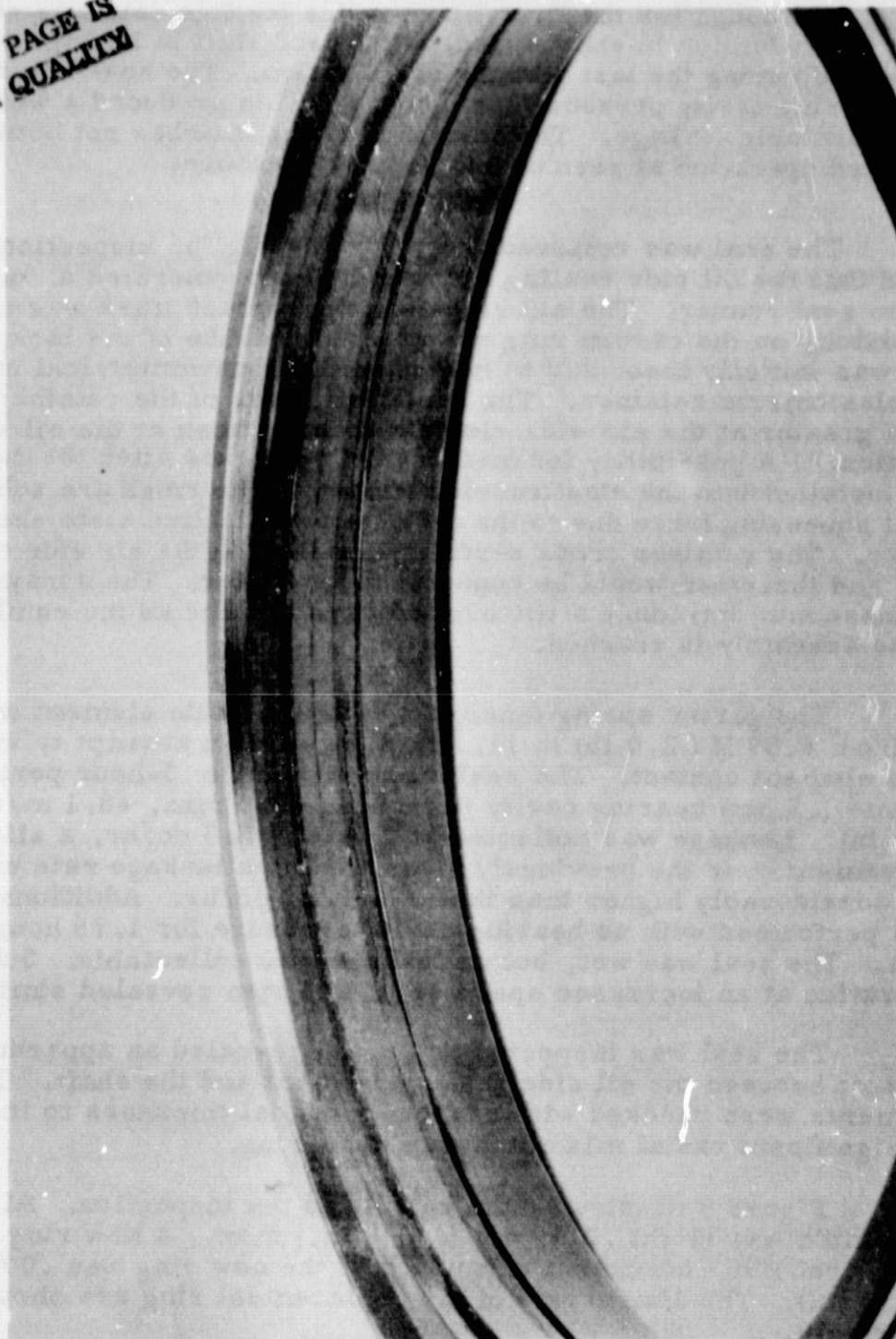


Figure 8. Seal Inside Diameter Before Test.

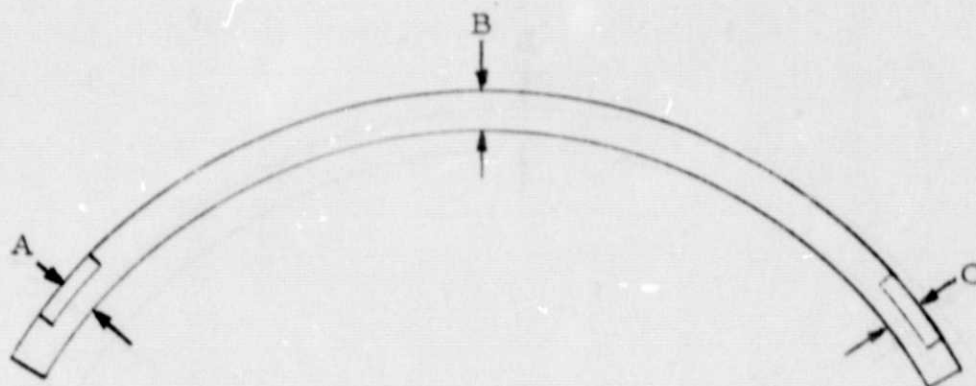
Although for the first 9 hours, the leakage rate was reduced with the reduction in shaft speed, an upward shift in leakage rate was observed during the last 3 hours of operation. The seal was rerun without bearing cavity pressure for 2 hours. This produced a wet seal but no collectable leakage. The shift in performance has not noticeably affected operation at zero bearing cavity pressure.

The seal was removed from the test rig for inspection. It was found that the oil side sealing element had not generated a contact mark on the seal runner. The air side element contact mark was obvious and burnishing on the carbon ring evident. The cause of the lack of contact was initially theorized to be due to the nonsymmetrical nature of the elastomeric retainer. The radial stiffness of the retainer appeared to be greater at the air side element location than at the oil side element location. A possibility for mispositioning occurs after the carbon rings are installed into the elastomeric retainer. The rings are subjected to an axial squeezing force due to the deflection of the free state shape of the retainer. The retainer cross section is thicker on the air side than on the oil side and therefore would be expected to be stiffer. The nonsymmetrical stiffness may introduce a tilt into the carbon rings as the equilibrium position of the assembly is reached.

The garter spring tension over the oil side element was increased from 8.89 N (2.0 lb) to 11.1 N (2.5 lb) in an attempt to improve oil side element contact. The seal was rerun for a 3-hour period at .34 N/cm² (.5 psi) bearing cavity pressure, 6600 rpm, 48.1 m/s (9,470 ft/min). Leakage was collected at a rate of 8.3 cc/hr, a slight improvement over the previously measured high leakage rate of 9.3 cc/hr, but considerably higher than the low of 2.1 cc/hr. Additional testing was performed with no bearing cavity pressure for 1.75 hours at 6600 rpm. The seal was wet, but no leakage was collectable. Subsequent operation at an increased speed of 10,000 rpm revealed similar results.

The seal was inspected and again revealed an apparent lack of contact between the oil side sealing element and the shaft. The carbon elements were checked with respect to radial thickness to insure that no significant radial mismatch was occurring.

Figure 9 illustrates the results of the inspection. Although the mismatch was slight .0119 cm (.0047 in.) max., a new ring was used on the rebuild. Maximum mismatch of the new ring was .0076 cm (.0030 in.). The dimensions of the replacement ring are shown in Figure 10.

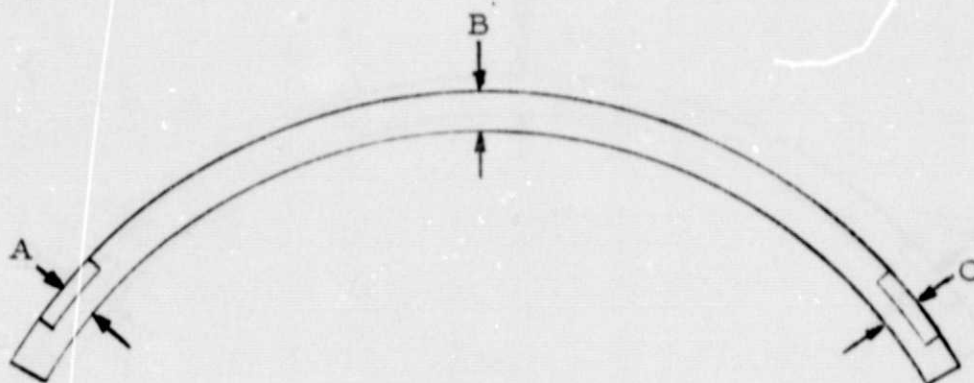


Element A (Air Side)	Location					
	A		B		C	
	cm	(in.)	cm	(in.)	cm	(in.)
Segment 1	.4180	(.1648)	.4148	(.1633)	.4189	(.1649)
Segment 2	.4171	(.1642)	.4171	(.1642)	.4115	(.1620)
Segment 3	.4110	(.1618)*	.4178	(.1645)	.4173	(.1643)

Element C (Oil Side)						
Segment 1	.4120	(.1622)	.4135	(.1628)	.4166	(.1640)
Segment 2	.4229	(.1665)*	.4117	(.1621)	.4209	(.1657)
Segment 3	.4168	(.1641)	.4150	(.1634)	.4166	(.1640)

* Indicates High and Low Elements

Figure 9. Carbon Element Radial Dimensions (Disassembly).



Element A (Air Side)

	Location					
	A		B		C	
	cm	(in.)	cm	(in.)	cm	(in.)
Segment 1	.4180	(.1648)	.4148	(.1633)	.4189	(.1649)*
Segment 2	.4171	(.1642)	.4171	(.1642)	.4115	(.1620)
Segment 3	.4110	(.1618)	.4178	(.1645)	.4173	(.1643)

Element E (Oil Side)

Segment 1	.4150	(.1634)	.4148	(.1633)	.4163	(.1639)
Segment 2	.4181	(.1646)	.4112	(.1619)*	.4161	(.1638)
Segment 3	.4168	(.1641)	.4161	(.1638)	.4186	(.1648)

*Indicates High and Low Elements

Figure 10. Carbon Element Radial Dimensions (Rebuild)

For the next test, in an attempt to achieve contact of the oil side element the garter spring of the air side element was removed. The conditions were 6600 rpm at $.34 \text{ N/cm}^2$ (.5 psi) bearing cavity pressure for 1.75 hours. The leakage rate was 11.7 cc/hr (8.3 cc/hr previously). Bearing cavity pressure was reduced to zero, and two more hours of operation yielded a leakage rate of .5 cc/hr. (none collectable previously). No evidence of oil side element contact was seen.

Summary of Initial Test Results

Testing thus far has been accomplished on a seal configuration varying only garter spring loads and substituting within the seal assembly essentially identical seal components. The seal functioned at shaft speeds up to 10,000 rpm, 72.9 m/s (14,349 ft/min) with acceptable leakage levels of .5cc/hr or less. This was achieved with no bearing cavity pressure. Operation with positive bearing cavity pressures produced significant leakage rates.

A lack of oil side element contact with the shaft was observed, and remedies were attempted by changing garter spring loads and the oil side carbon sealing element. None of the modifications were successful in improving sealing element contact or reducing oil leakage. The sudden increase in leakage to excessive rates during the 6600 rpm test run was not explained. Elastomeric swelling may be the cause.

Modifications and Subsequent Testing

Further testing was carried out with modifications to the sealing element. A discussion of these modifications and tests follows.

Anti-Rotation Lug Rework

Stretching of the elastomeric retainer around each oil side anti-rotation lug was observed. Since this condition with elastomer swelling might produce a radial unloading effect with respect to the sealing element, the anti-rotation lugs of the oil side element were reworked by removing material (.064 cm (.025 in.)) from their underside (Figure 11). This resulted in less stretching of the retainer. A 2-hour retest of the seal with this modification at 6600 rpm and $.34 \text{ N/cm}^2$ (.5 psi) bearing cavity pressure yielded a leakage rate of 24 cc/hr (11.7 cc/hr prior to rework). The test was carried out with a single garter spring over the oil side element. No evidence of oil side element contact was seen.

Ramp Rework

A 45-degree ramp approximately one-half the radial thickness

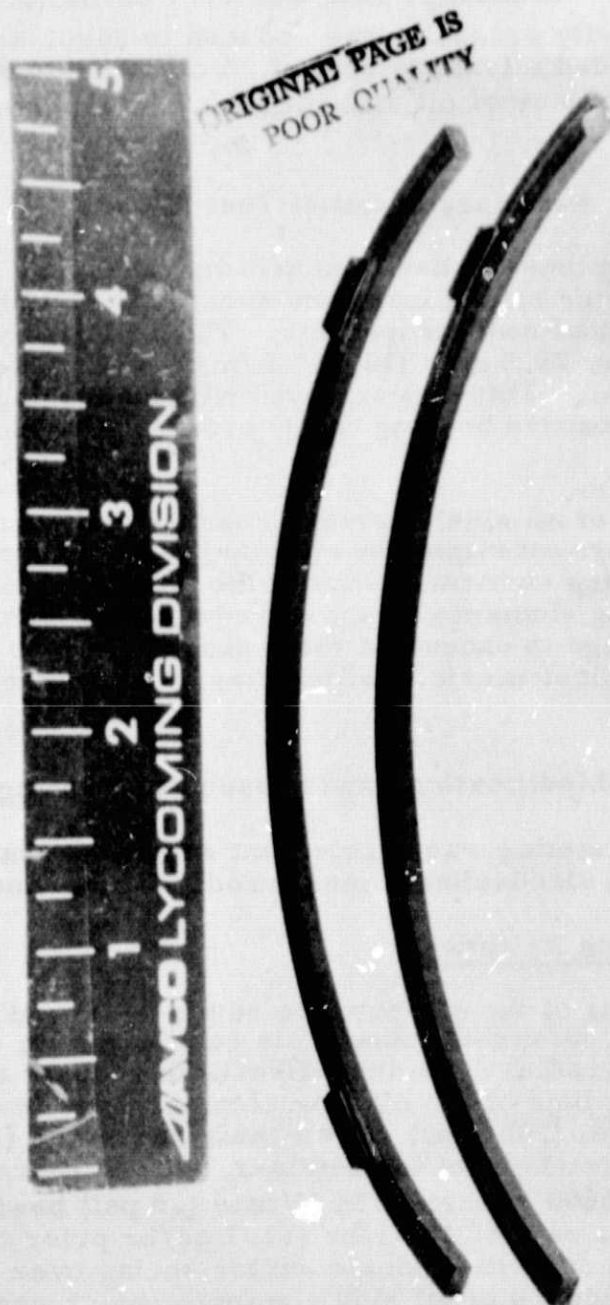


Figure 11. Carbon Element Modifications.

of the sealing element was added to the leading end of each oil side segment (Figure 11). The intent of this modification was to force back into the bearing cavity oil which entered the segment end gaps. Since no improvement in performance had been observed during testing with a single garter spring over the oil side element, both springs were in place during this test. The seal, which was tested at 6600 rpm and $.34 \text{ N/cm}^2$ (.5 psi), exhibited a leakage rate of 30 cc/hr. The seal was retested at no bearing cavity pressure at 6600 rpm for 2.75 hours and revealed a leakage rate of 3.45 cc/hr.

Trailing Anti-Rotation Lug Rework

The trailing anti-rotation lugs of the oil side sealing element were removed in an attempt to provide the element with additional radial freedom. The seal was tested for 3 Hours at 6600 rpm, $.34 \text{ N/cm}^2$ (.5 psi) with a leakage rate of 22 cc/hr. With the bearing cavity pressure reduced to zero, the leakage rate dropped to 18.6 cc/hr. Figure 11 illustrates by comparison all the modifications incorporated into the oil side carbon elements. These modifications were not successful in reducing leakage nor improving oil side carbon element contact with the seal runner.

A test was performed with a new elastomeric retainer in an attempt to identify what effect, if any, the retainer was having on oil leakage. Knowing that elastomers swell slightly as a function of their exposure to oil provided a rationale for suspecting that some of the leakage effect may be due to the retainer. No actual swell measurements were attempted. The leakage rate at 6600 rpm and $.34 \text{ N/cm}^2$ (.5 psi) was 16 cc/hr. At 0.0 N/cm^2 , the leakage rate dropped to 2.3 cc/hr. This was an improvement from the previous levels measured and indicates that the retainer can affect leakage and that elastomeric swelling may have been significant factor. The retainer that was replaced is shown in Figure 12 (43.75 hours operating time).

The cooling oil flow was redirected to impinge directly on the inside diameter of the steel runner and was increased from 83.9 kg/hr (185 lb/hr) to 136 kg/hr (300 lb/hr) to increase the cooling effect with the intent to reduce any temperature associated distortions of the seal runner. Measured seal leakage remained unchanged; however, the higher oil flow level was maintained throughout the remaining tests. The seal was removed and inspection revealed an extremely slight edge contact on the oil side element and typical contact on the air side element. Figure 13 illustrates the assumed positioning of the carbon rings necessary to produce the observed contact indications. This type of contact would produce a direct leakage path. Photographs of the air side and oil side elements are shown in Figures 14 and 15.

The seal was returned to its original configuration by replacing the oil side element with an unmodified ring. Testing at 6600 rpm and $.34 \text{ N/cm}^2$ (.5 psi) bearing cavity pressure yielded a leakage rate of



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Figure 12. Replaced Retainer .

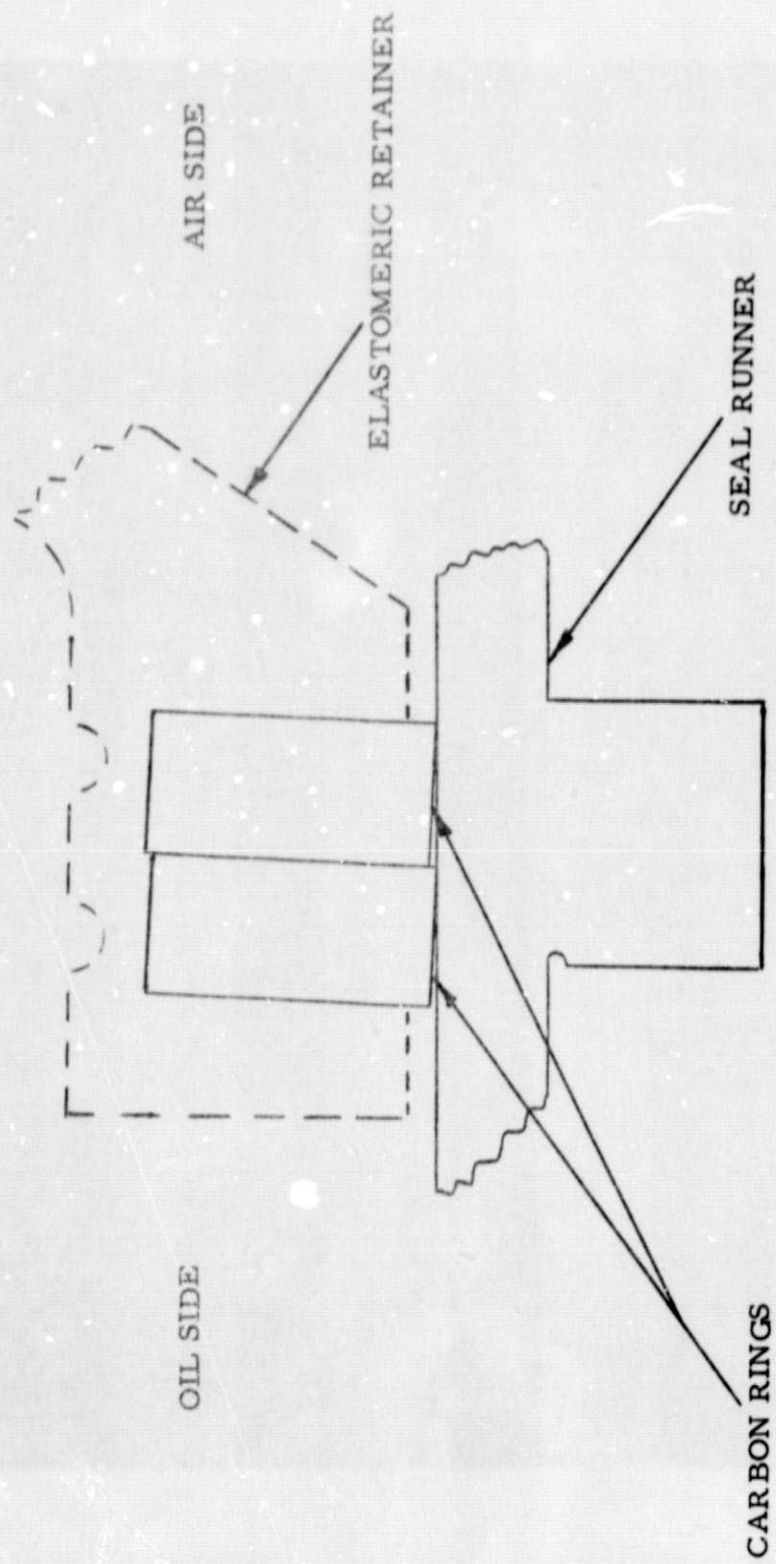


Figure 13. Mispositioned Carbon Rings.

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Figure 14. Air Side Elements After Test.

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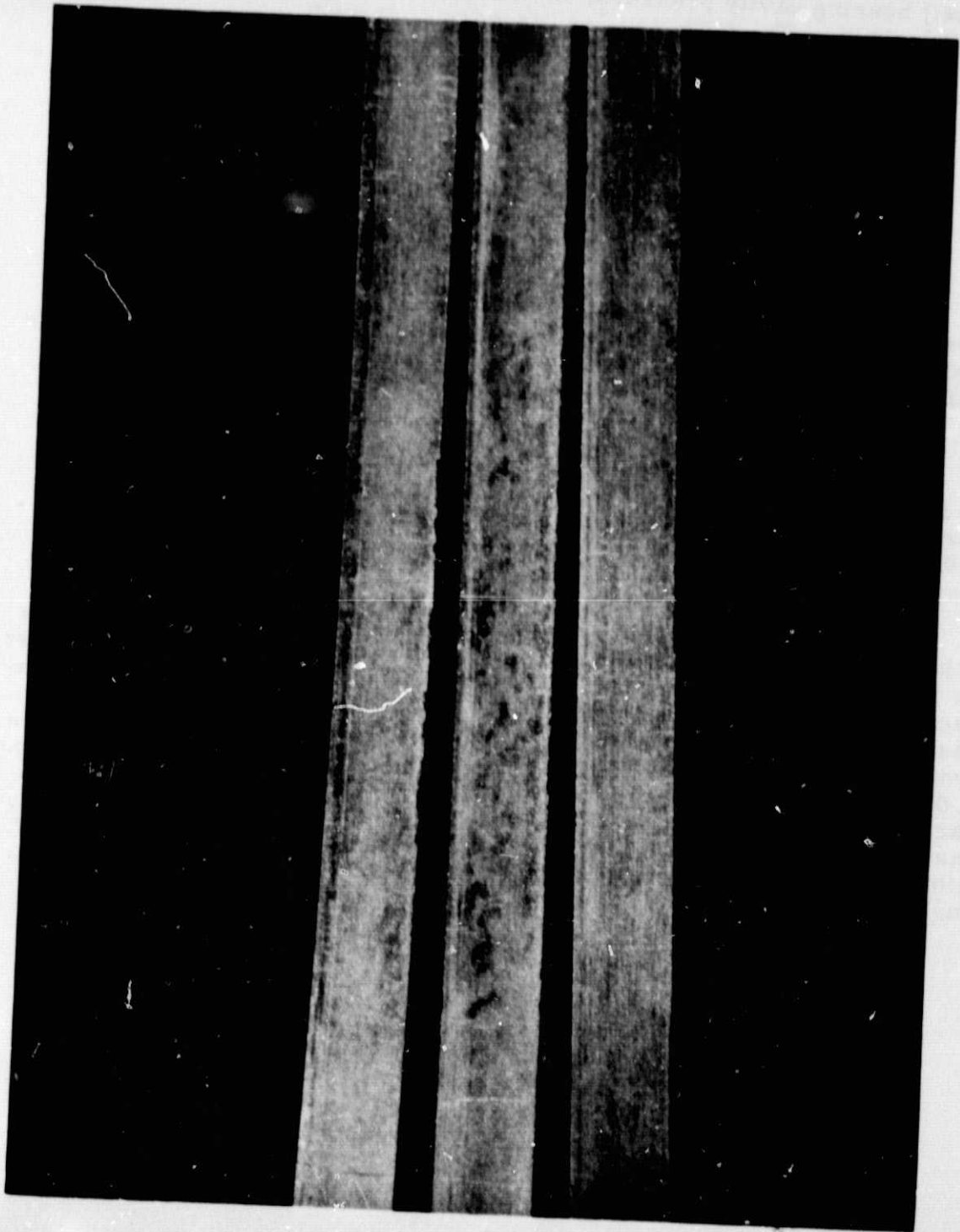


Figure 15. Oil Side Elements After Test
(With Modifications).

26 cc/hr. Reducing bearing cavity pressure to zero reduced leakage to 1.23 cc/hr. The test rig speed was reduced, maintaining .34 N/cm² (.5 psi) bearing cavity pressure, in order to determine the effect of speed on oil leakage. The following results were observed.

<u>RPM</u>	<u>Leakage Rate</u>
3300	0.0 cc/hr
4500	32 cc/hr
3700	0.0 cc/hr

The leakage rate exhibited essentially a step change at approximately 4500 rpm, where a high leakage rate was observed. The implication of this performance is that the seal may be unable to follow shaft motion, thus opening a leakage path across the bore.

Seal torque measured over the test speed range is presented in Figure 16. As can be seen, seal torque decreases with speed and varies insignificantly with changes in bearing cavity pressure between 0.0 and .34 N/cm² (.5 psig). This concluded the planned 65 hours of testing.

The seal was removed from the test rig, and photographs were taken of the seal components. Figure 17 illustrates the seal and runner at the completion of testing. A close-up of the seal runner and the contact area can be seen in Figure 18. An axial trace of the outside diameter of the seal runner shown in Figure 19 illustrates the depth of the worn area. Maximum wear of the carbon rings was observed at .00025 cm (.0010 in). A close-up of the seal bore in Figure 20 illustrates a contact (burnishing) on the air side sealing element extending for approximately 60 percent of the ring width. The oil side sealing element exhibits only very slight contact at the edge adjacent to the air side sealing element.

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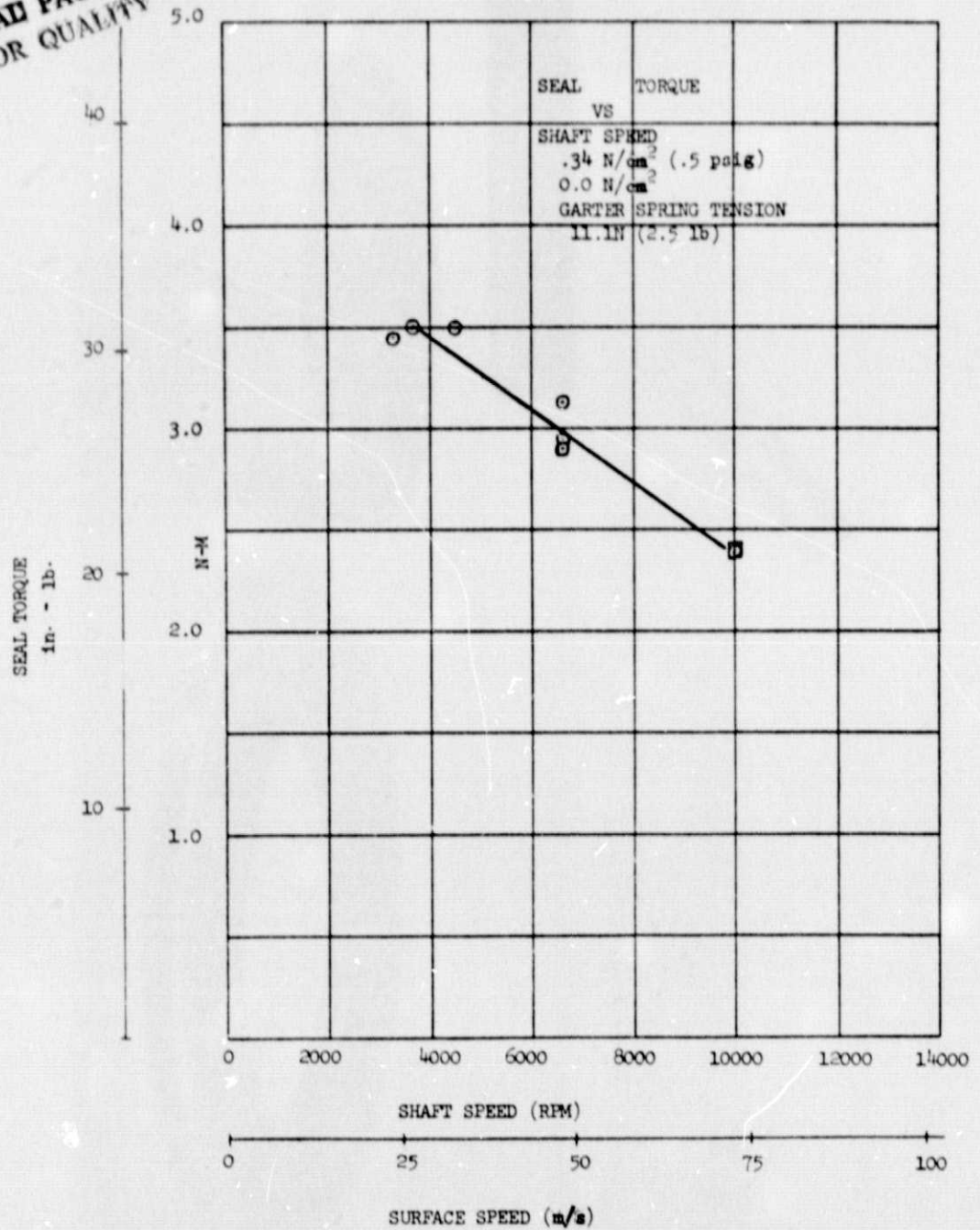


Figure 16. Seal Torque Versus Speed.

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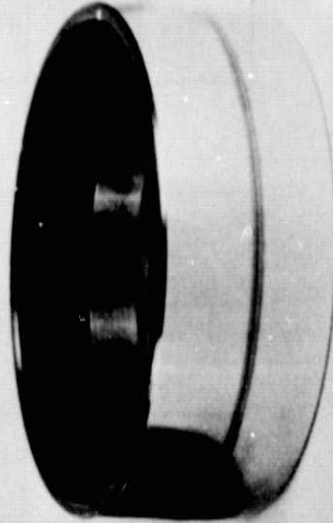


Figure 17. Seal and Runner After Test.

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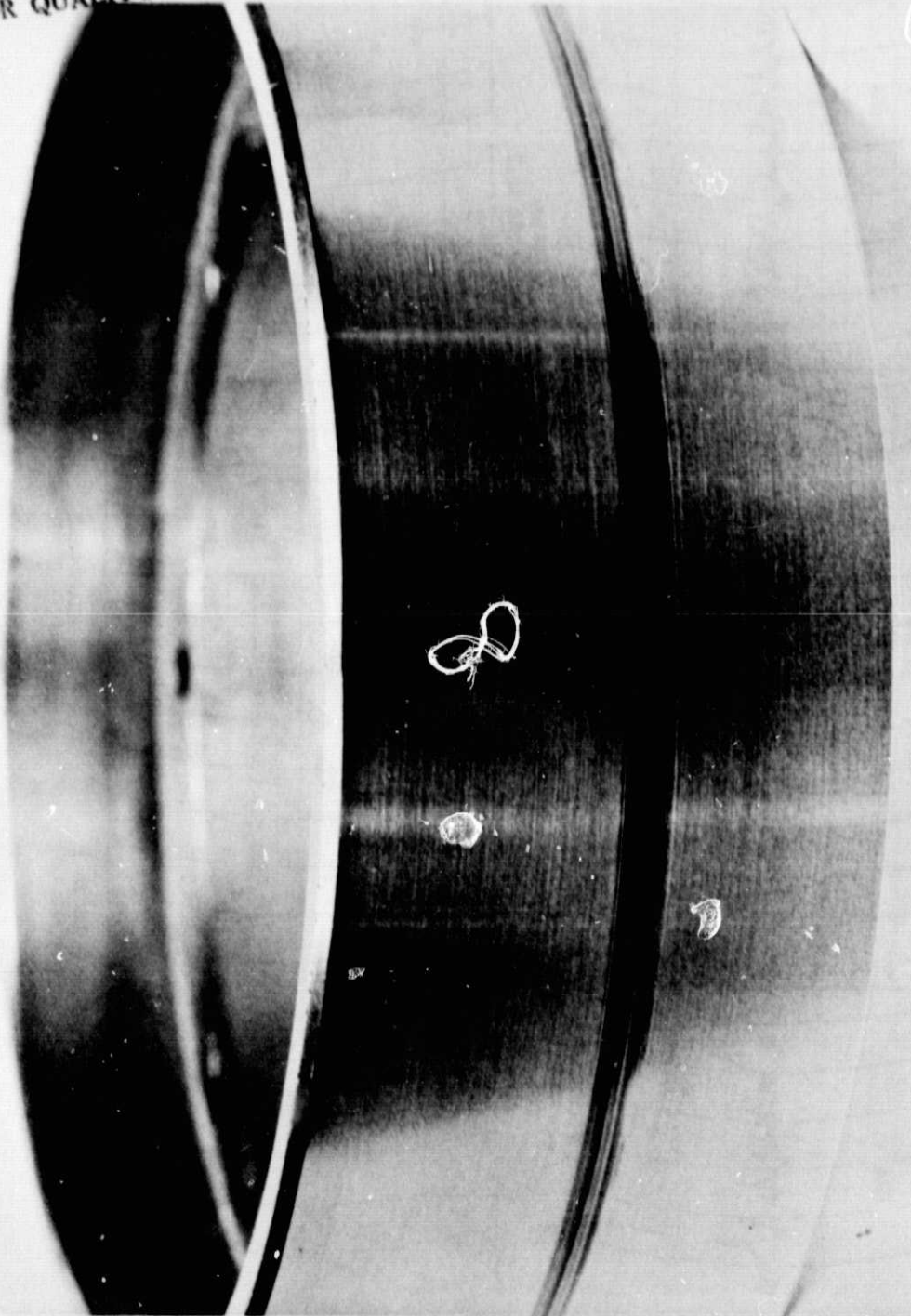


Figure 18. Seal Runner After Test.

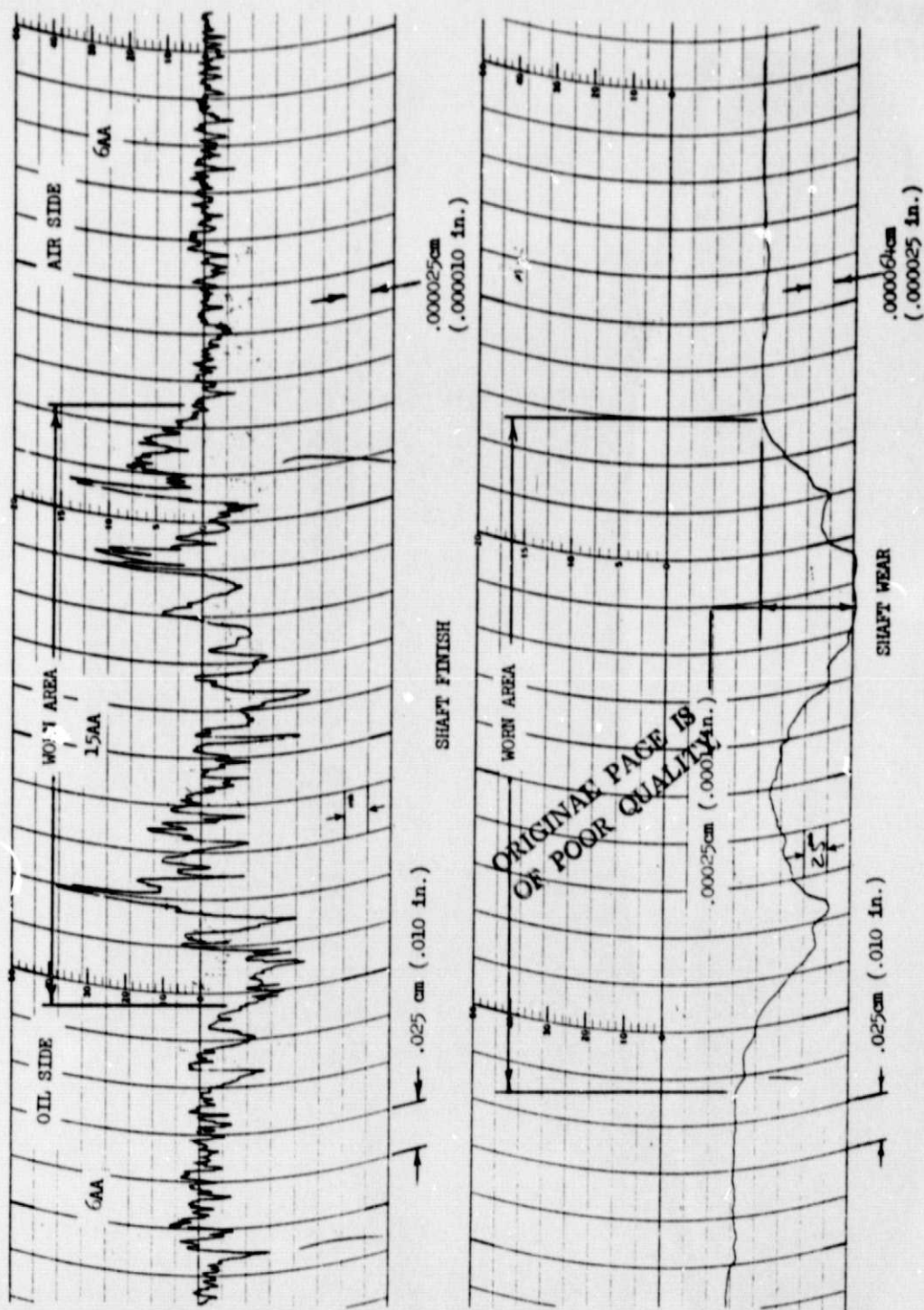


Figure 19. Seal Runner Finish and Wear After Test.

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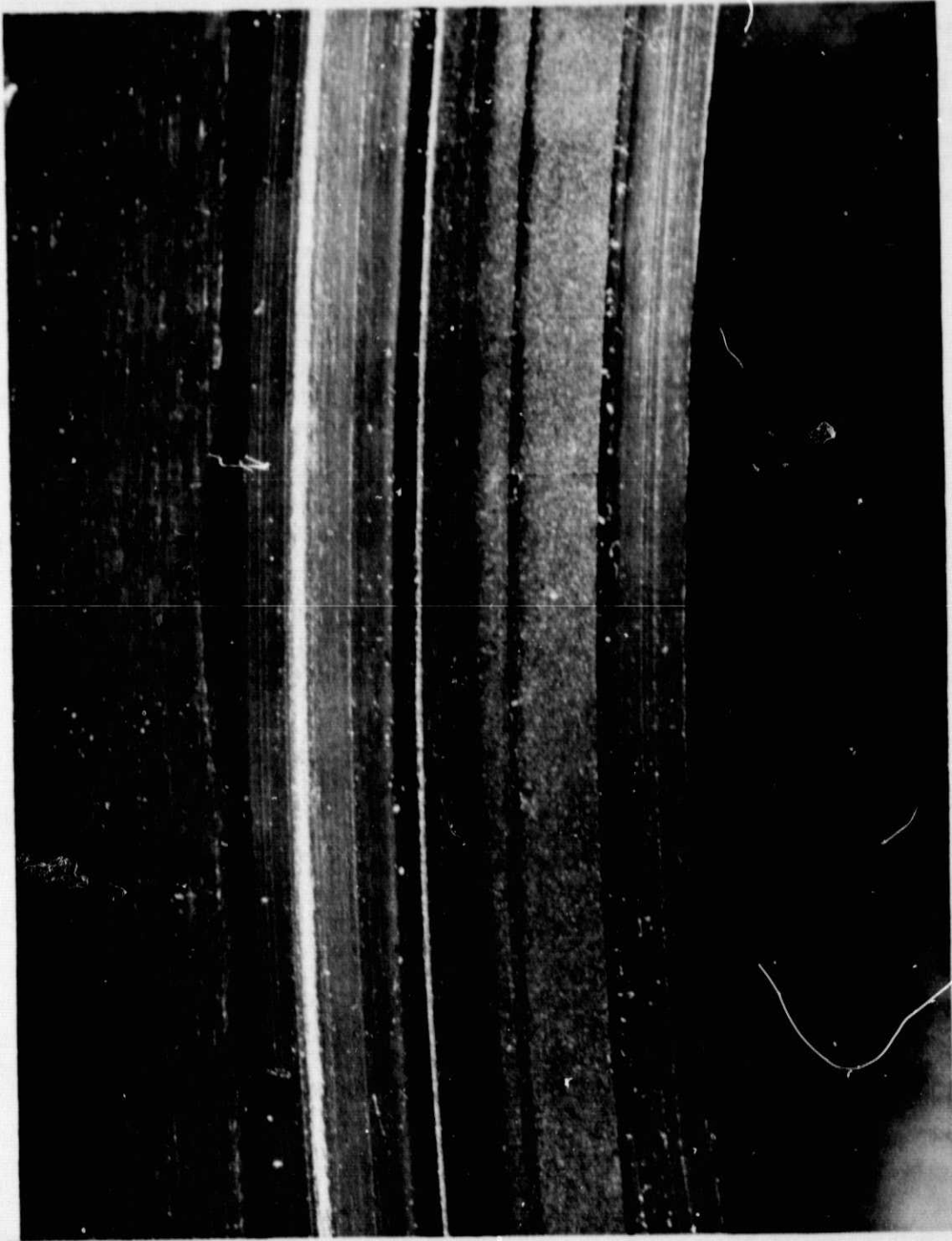


Figure 20. Seal Bore After Test.

DISCUSSION OF RESULTS

Initial testing revealed that the synergistic seal exhibited acceptable oil leakage rates (.47 cc/hr) at speeds up to 10,000 rpm, 72.9 m/s (14,349 ft/min) with 0.0 N/cm² bearing cavity pressure. When tests were conducted with .34 N/cm² (.5 psi) bearing cavity pressure leakage rates were higher and unstable, even at reduced speeds (9.3 cc/hr at 6600 rpm). Later testing at still lower speeds (3700 rpm) revealed a reduction in leakage producing a wet seal but no collectable amount of leakage.

Potential causes of oil leakage are as follows:

1. Problems associated with the nonsymmetrical nature of the elastomeric retainer manifested as apparent lack of contact between the oil side sealing element and the shaft and mispositioning or tilting of the sealing elements with respect to the shaft - Suspected because of the character of observed wear tracks.
2. Possible inability of the sealing element to follow the shaft - Concluded from the increase in oil leakage rate with increasing speed.
3. Elastomeric retainer dimensional changes after exposure to oil - Suspected because of the observed leakage decrease after the replacement of the original, undamaged, elastomeric retainer.

Modifications incorporated into the carbon sealing elements in an effort improve oil side sealing elements shaft contact produced unsuccessful results. Improving the symmetry of the elastomeric retainer to minimize the nonsymmetrical effects suspected of causing oil leakage should be beneficial. An alternate approach to minimizing these effects would involve a final bore honing of the sealing rings after assembly into the elastomeric retainer, thereby re-establishing the sealing bore.

Although elastomeric retainer swell is identified as a cause of leakage, the actual interaction of the elastomeric retainer and the other seal components causing increased leakage rates is not yet understood. Elastomeric swell may increase retainer nonsymmetry.

Because of its similarity to a lip seal in structure the synergistic seal may be expected to be subject to similar limiting factors. High-speed lip seals tend to be limited by combinations of shaft runout and misalignment between the seal and shaft geometric centers and heat generation at the lip. Although gains in surface speed capability can be reasonably expected by virtue of the use of carbon-graphite instead of an elastomer as a rubbing material, a dynamic shaft tracking limitations still exist. Longer flex sections are often used to improve runout and speed capability of lip seals. The synergistic seal, having a relatively thick, short flex section, could benefit in shaft following ability by lengthening and thinning out this section (Figure 21).

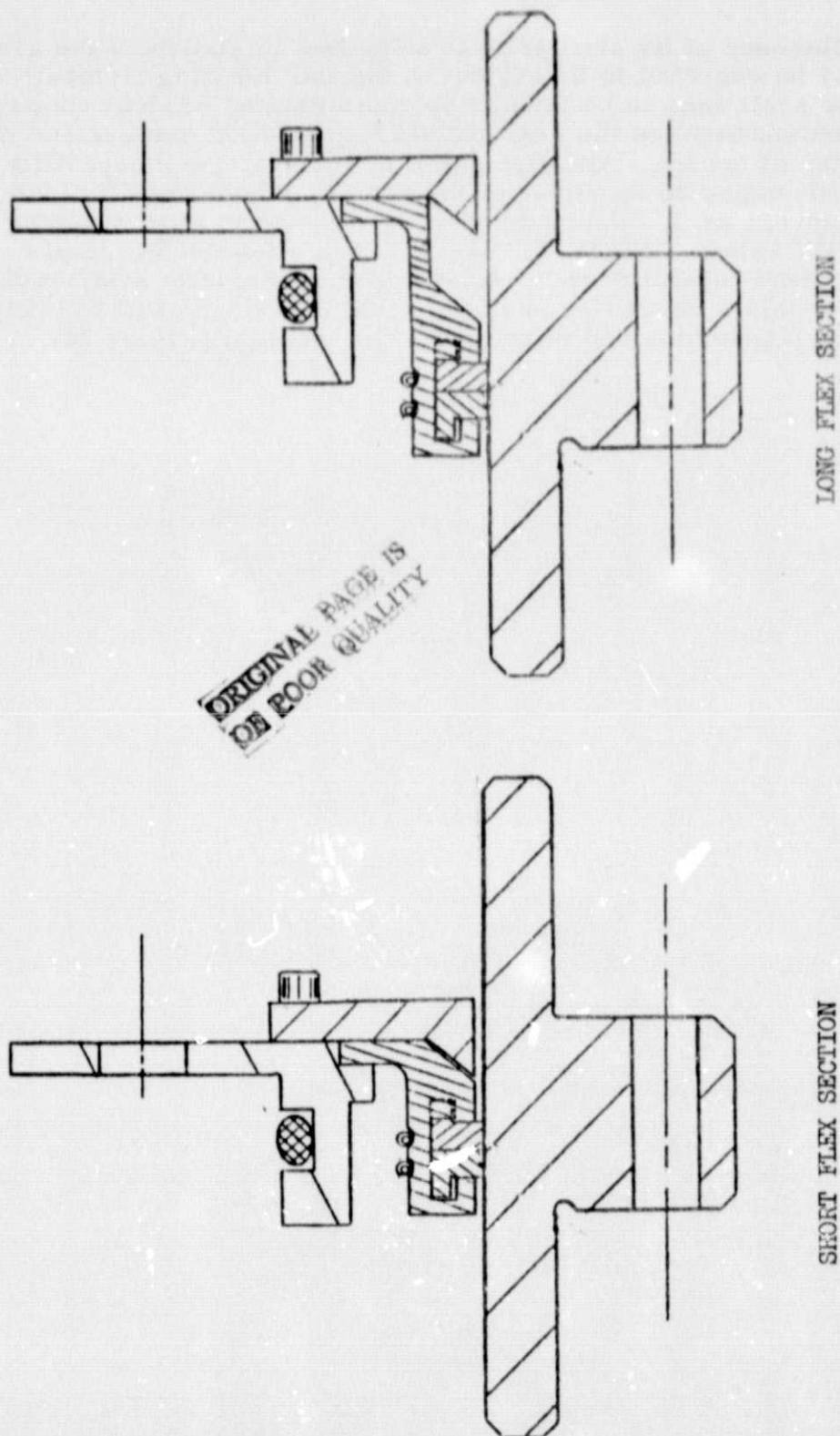


Figure 21. Short Versus Long Flex Section.

CONCLUSION

The synergistic seal requires additional development before it can be considered for use in advanced aircraft or helicopter transmissions. Although this test program has demonstrated solutions to problems identified in prior programs (anti-rotation lug failures in Reference 1), seal oil leakage at positive bearing cavity pressures remains a problem area.

The specific cause of the oil leakage has not been determined. The most likely causes can be summarized as follows:

The oil side sealing element makes very limited or at times no contact with the shaft, causing a direct leakage path.

The wear indications on the sealing element bores show the sealing rings are operating tilted with respect to the seal runner, causing a direct leakage path.

The mechanism causing these conditions is believed to be related to the nonsymmetrical shape of the elastomeric retainer.

An additional leakage effect may be due to shaft tracking limitations associated with the elastomeric retainer as it is presently configured.